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THE EFFECT OF UNBALANCED VERGE TORQUES, ACCELERATION FRICTION TORQUES, AND VERGE-STARWHEEL MATERIAL UPON THE RUNNING RATES OF A CERTAIN TYPE OF TIME DELAY MECHANISM

A REPORT OF WORK CARRIED OUT UNDER CONTRACT
DAI-49-186-502-ORD(P)-186 FOR THE
DIAMOND ORDNANCE FUZE LABORATORIES

VIRGINIA MILITARY INSTITUTE
DEPARTMENT OF PHYSICS
LEXINGTON, VIRGINIA
JANUARY 1955

ABSTRACT OF REPORT

Three factors not hitherto considered will effect the running rate of a certain type of time delay mechanism.

(1) If the verge is not balanced, when it is accelerated an unbalanced torque is set up Under extreme conditions of acceleration and unbalance the running rate would not be changed by more than a few percent. The mechanism will run faster if the unbalanced torque is applied to trailing side of verge, slower for torque on leading side, at least until certain jamming values of torque are reached.

(2) Friction has been neglected in our previous reports. As the mechanism accelerates this friction torque is increased. There was no agreement between theory and experiment. All experimental results indicate an <u>increase</u> in the running rate as friction torque increases up to the jamming point.

(3) If the material of the verge and starwheel is changed, the running rate will change, but probably not more than 5% from the value for brass.

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The Effect of Unbalanced Verge Torques, Acceleration
Friction Torques, and Verge-Starwheel Material Upon
The Running Rates of a Certain Type of Time
Delay Mechanism

I. INTRODUCTION

The time delay mechanism has been studied in detail under this and a
preceding contract. Previous results are embodied in technical reports made to
Diamond Ordnance Fuze Laboratories, and deposited in the library. These technical
reports are three in number and reference to them will be made by symbols R_1, R_2,
and R_3. The mechanism is described in detail in each of these three reports
and later in this report a partial description will be given. All previous
studies have dealt with the behaviour of the mechanism while it was at rest. It
is of course obvious that when inserted into a shell of any kind, such a time de-
lay mechanism must operate while accelerating rapidly. Since the mechanism
consists largely of gears and a verge each supported on a shaft passing through
holes in a metal casing, friction torques due to such acceleration will appear.
Additionally, if any wheel or the verge is dynamically out of balance, then a
torque will be set up. Either or both of these effects may radically alter the
behaviour of the mechanism and also its running time.

Finally, it is quite possible that a change in the material of which
the verge, the starwheel, or both, is made might so change the character of the
motion as to introduce a definite change in running time.

II. UNBALANCED TORQUES - THEORY

It is necessary first to investigate theoretically the effect of a dynamically unbalanced verge or starwheel upon the motion of the combination. As in previous work, a scaled up model was the basis of investigation and theory deals only with this model. Complete geometrical characteristics of the motion are given in R_1 - II, R_3 - I.

It is true that unbalanced torques introduced into the gear train preceding the starwheel will affect the running time, through their effect upon the torque applied to the starwheel. We will for the present be concerned only with an unbalanced torque on the verge itself, since any unbalanced constant torque on the starwheel can only increase or diminish that constant applied torque already present. We have already found how a change in T effects $\bar{\theta}$ We consider only that case for which the acceleration is a constant, since this can be reproduced experimentally.

Reference to R_1 - II will supply many details omitted in the following discussion. Acting on the wheel is a constant torque T and the reaction torque $F \cdot u$, and acting on the verge are the torque $F'v$ and an unbalanced torque T'. We have considered T' as a constant although this is not strictly true since the lever arm of this torque varies with the cosine of the angle of the verge. Our past experience indicates that the verge angle will never exceed a rotation of 5° in either direction from equilibrium and so the cosine will change by less than 0.5% Hence our assumption of constant lever arm seems justified.

Supposing that the mechanism accelerates as shown by the arrow then this unbalanced torque could be considered as $T' = MAL$, where P is the center of mass. Of course there is no basic reason to suppose that the mass center is displaced left or right of the axis of rotation rather than up or down or any other direction.
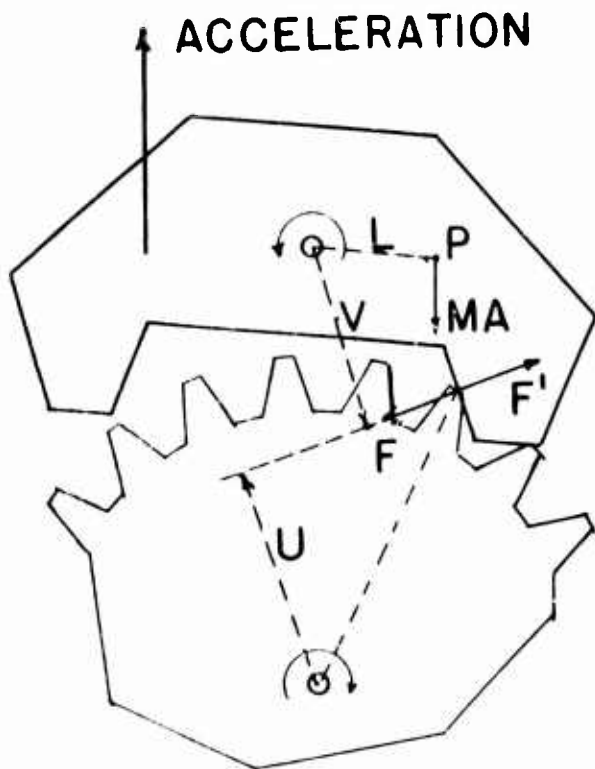

ACCELERATION

FIG. 1

However it is true that the maximum torque τ' will result for the displacement as shown.

The differential equations of motion are then:

Verge: $\tau' + F'v = I_v \ddot{\alpha}$

Wheel: $\tau - F \cdot u = I_w \ddot{\theta}$

Where τ' is here a negative number, since its direction is clockwise.

Dividing: $\dfrac{-F \cdot u}{F'v} = \dfrac{I_w \ddot{\theta} - \tau}{I_v \ddot{\alpha} - \tau'}$

Since $F = F'$, $u/v = \dfrac{d\theta}{d\alpha}$ we have

$$(I_v \ddot{\alpha} - \tau')d\alpha = (I_w \ddot{\theta} - \tau)d\theta$$

Integrating gives

$$\dot{\theta}^2 = \dfrac{\left[I_w + I_v\left(\frac{d\alpha}{d\theta}\right)^2\right]\dot{\theta}_0^2 + 2\tau(\theta - \theta_0) + 2\tau'(\alpha - \alpha_0)}{\left[I_w + I_v\left(\frac{d\alpha}{d\theta}\right)_0^2\right]}$$

for the motion of the wheel during the time it is contact with the verge. θ_0, $\dot{\theta}_0$ are position and velocity of wheel at the beginning of angular interval. Remember that a clockwise rotation of the wheel is positive, but a counter clockwise rotation of verge is positive.

A similar discussion, with particular attention paid to signs leads to exactly the same equation for contact on the trailing face. Note however that $(\alpha - \alpha_0)$ is now of opposite sign.

Now suppose the mass center lies to the left rather than to the right of the axis of rotation. τ' is here positive. The net result is to obtain the same equation as has already been given above, so that this equation can be used in all situations. Now Appendix A of R_2 explains precisely how the complete cycle may be solved for $\dot{\theta}$ as a function of θ, with the exception that a term $2\tau'(\alpha - \alpha_0)$ must be added in the numerator of equations (1) and (5) for leading unbalanced torque, and subtracted for the case of trailing unbalanced torque.

III. THEORETICAL RESULTS - UNBALANCED TORQUES

The equations mentioned above were applied to 8 verge-wheel combinations as follows.

$$I_v = 2085 \text{ gm. cm.}^2 \qquad\qquad I_w = 2100 \text{ gm. cm.}^2$$

$$\mathcal{T} = 2.71 \times 10^6 \text{ dyne cm.} \qquad \mathcal{T}' = 0, \pm 8.68 \times 10^4, \pm 5 \times 10^5 \text{ dyne cm.}$$

$$\mathcal{T} = 1.08 \times 10^6 \text{ dyne cm.} \qquad \mathcal{T}' = 0, \pm 8.68 \times 10^4 \text{ dyne cm.}$$

We found that where $\mathcal{T}' < 0$ (torque applied on leading side of the verge) that $\dot{\Theta}$ was smaller than for the case $\mathcal{T}' > 0$ (applied on trailing side). The cycles of motion are not drawn here as computed, but Fig. 2 indicates in exaggerated form the relative values obtained in e cases $\mathcal{T} = 2.71 \times 10^6$ dyne cm.

$$\mathcal{T}' = 0, \pm 8.68 \times 10^4 \text{ dyne cm}, \pm 5 \times 10^5 \text{ dyne cm}$$

The values of $\dot{\Theta}$ in each of these cases are given below.

$$\mathcal{T}' = 0$$
$$7.93 \text{ rad/sec}$$

$$\mathcal{T}' = 8.68 \times 10^4 \text{ dyne cm} \qquad\qquad \mathcal{T}' = -8.68 \times 10^4 \text{ dyne cm}$$
$$8.05 \text{ rad/sec} \qquad\qquad\qquad\qquad 7.73 \text{ rad/sec}$$

$$\mathcal{T}' = +5 \times 10^5 \text{ dyne cm} \qquad\qquad \mathcal{T}' = -5 \times 10^5 \text{ dyne cm}$$
$$8.19 \text{ rad/sec} \qquad\qquad\qquad\qquad 7.51 \text{ rad/sec}$$

When $\mathcal{T} = 1.08 \times 10^6$ dyne cm, theoretical results are

$$\mathcal{T}' = 0$$
$$(\text{not obtained})$$
$$\mathcal{T}' = 8.68 \times 10^4 \text{ dyne cm} \qquad\qquad \mathcal{T}' = -8.68 \times 10^4 \text{ dyne cm}$$
$$5.16 \text{ rad/sec} \qquad\qquad\qquad\qquad 4.92 \text{ rad/sec}$$

These figures indicate that as an increasing positive torque is applied $\dot{\Theta}$ increases. Of course in each ca e once $\mathcal{T}'$ becomes large enough the motion must cease and $\dot{\Theta}$ becomes zero.

Certain qualitative physical analyses may be applied to these cycles which anticipate these results. Consider first the case in which the torque is positive, that is, on the trailing side. Reference to Figure 1 will help understand what follows. All velocities and collision angles are compared with what they would be in the normal case, in which the unbalanced torque is zero. Suppose the starwheel is just making last contact leading with the verge. As the verge and wheel go into the leading free motion, the verge moves more rapidly than normal, so comes around further before trailing collision. This means that the wheel turns a smaller distance before trailing collision. The free period

UNBALANCED TORQUE CYCLES (EXAGGERATED)

$\tau = 2.71 \times 10^6$

$T' = 0$

$T'_1 = +8.68 \times 10^4$ $T'_1 = 8.68 \times 10^4$
$T'_1 = 15.00 \times 10^5$ $T'_1 = -5.00 \times 10^5$

TORQUES IN DYNE-CM

WHEEL VELOCITY, RAD/SEC.

WHEEL POSITION, RADIANS

FIG. 2
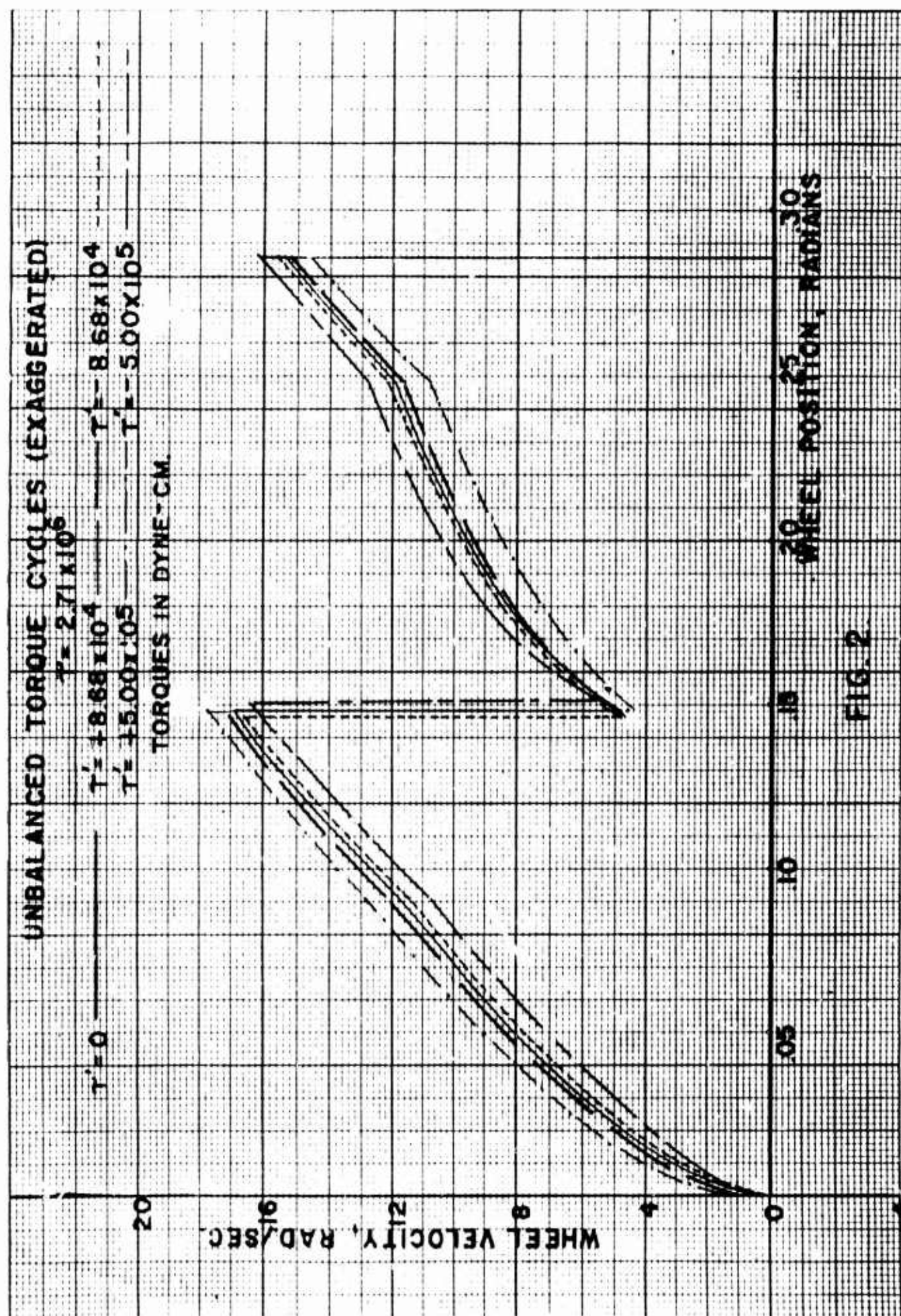
lasts a shorter time, but the velocity of verge is greater when collision occurs, while that of the wheel is smaller. Now the angular duration of trailing contact motion is increased and at the same time the unbalanced torque opposes the wheel motion during this interval, so its velocity is smaller than normal. After last contact trailing the verge slows down because of the unbalanced torque and hende the angular duration of the free motion of the wheel is greater than normal. After leading collision, the contact duration is smaller to last contact leading, but the unbalanced torque aids the verge and wheel to move. The overall result is easily seen to be that during the leading half of cycle there is an increased velocity and for trailing half a decreased velocity, this occuring when the unbalanced torque is positive, on trailing side.

One can deduce analogous results for the negative unbalanced torques in which the velocity of the wheel is decreased for loading half of cycle and increased for trailing half. These effects show clearly on the graph of Figure 2. Unfortunately it is not possible to predict the overall affect on a cycle from these qualitative considerations. It turns out, as our results show, that for a cycle in which the velocity is greater on the leading half than normal, so will the average velocity be greater than normal, and vice versa. We shall meet this type of correlation again, and, in fact, have noticed it before in previous work. It seems to be true that the average velocity of a cycle is fairly well correlated with the behaviour of the mechanism during the leading half.

Repeating our conclusions, an unbalanced torque on the trailing side speeds up the motion of the mechanism and one on leading side slows it down. Are these theoretical results borne out by experiment? A model verge of $I_v = 2085$ gm cm^2 but with an unbalanced mass was fabricated. By shifting a cylindrical plug from one side to the other it was possible to apply an unbalanced torque on either side without changing the moment of inertia. The results are:

for $\tau = 2.71 \times 10^6$ dyne cm

$\tau' = 8.68 \times 10^4$ dyne cm $\quad \tau' = 0 \quad \tau' = -8.68 \times 10^4$ dyne cm

$\bar{\dot\theta} = 7.70$ rad/sec $\quad \bar{\dot\theta} = 7.58 \quad \bar{\dot\theta} = 7.45$

for $\tau = 1.08 \times 10^6$ $\quad \tau' = 0 \quad \tau' = -8.68 \times 10^4$

$\tau' = +8.68 \times 10^4$ $\quad \bar{\dot\theta} = 4.85 \quad \bar{\dot\theta} = 4.66$

$\bar{\dot\theta} = 4.90$

As comparison with the theoretical values show, the results agree very well in the two cases. As always, experimental results are smaller than the theoretical because of friction.

IV. EXTRAPOLATION OF RESULTS TO MECHANISM

The results so far discussed are concerned with the behaviour of the model, which is scaled up 10 times from the actual mechanism. We have seen in R_1 - VIII that although we scale up the geometrical dimensions of the mechanism, we do not scale up the applied torques in the same ratio. The actual torque applied in the mechanism is approximately 5×10^2 dyne cm. Scaled up by a factor of 10^5 (to match moments of inertia) would necessitate $\tau = 5 \times 10^7$ dyne cm. We actually use $\tau = 2.5 \times 10^6$, so that we should be using torques some 20 times larger than we do.

We know from our measurements of R_3 - V that the maximum displacement of center of mass of verge off axis can hardly exceed .005 inches. The mass of the verge is about 0.19 dyne cm, or a value such that $\tau' = 0.4\tau$. This is of course an extreme value and it is likely that $\tau' = 0.2\tau$ would be much closer to a realistic value of τ'. Our theoretical results of the preceding paragraph show that for $\tau' = 0.2\tau$, the change of θ is no larger than 5%. We can perhaps safely extrapolate these results and say that within the probable limits of an unbalanced torque the change in velocity would certainly be less than 10%. We still have not considered the fact that the torque is some 20 times smaller than it should be for a true scaling up of all quantities. In our original work we found that the results obtained for the model did represent with good accuracy the behaviour of the actual mechanism. We conclude that probably no unbalanced torque we may expect to find in the actual mechanism will seriously change the running velocity of the mechanism.

V. THE EFFECT OF FRICTION TORQUES UPON THE RUNNING TIME - THEORETICAL

The basic equations which we need here can be derived very much as we found those in the case of unbalanced torques. The difference is one in direction of the friction torque as the motion of the verge changes. On the leading half of the cycle the verge moves in a counterclockwise or positive sense, so that the friction torque is negative. On the trailing half, the motion of the verge is in the negative sense, so that of the friction torque is positive. Hence the following equations will represent the motions on leading and trailing contact.

Leading
$$\dot{\Theta}^2 = \frac{\left[I_w + I_v\left(\frac{d\alpha}{d\theta}\right)_o^2\right]\dot{\Theta}_o^2 + 2\tau(\theta-\theta_o) - 2\tau'(\alpha-\alpha_o)}{\left[I_w + I_v\left(\frac{d\alpha}{d\theta}\right)^2\right]}$$

Trailing
$$\dot{\Theta}^2 = \frac{\left[I_w + I_v\left(\frac{d\alpha}{d\theta_o}\right)^2\right]\dot{\Theta}_o^2 + 2\tau(\theta-\theta_o) + 2\tau'(\alpha-\alpha_o')}{\left[I_w + I_v\left(\frac{d\alpha}{d\theta}\right)^2\right]}$$

We must remember that $(\alpha-\alpha_o) < 0$ on the trailing half. Considering $\tau' > 0$ always we see that the τ' term is in each case subtracted from the τ term. In such a cycle as this there can be a complication not hitherto met nor described. As τ' becomes larger, the verge, during either leading or trailing free motion, is more and more quickly brought to rest. If τ' is large enough, then the verge may possibly come to rest before collision with the wheel occurs. This will change the method of determining the angle of collision. The following equations are familiar, with the addition of terms involving τ' (see for instance, R_2 - Appendix A), and are used to find the trailing collision.

Verge   Wheel

(1) $\quad \alpha_2 = -\frac{\tau'}{2I_v}t^2 + \dot{\alpha}_1 t + \alpha_1$ (3) $\quad \theta_2 = \frac{\tau}{2I_w}t^2 + \dot{\theta}_1 t + (\theta_1 - \theta_k)$

(2) $\quad \dot{\alpha}_2 = -\frac{\tau'}{I_v}t + \dot{\alpha}_1$ (4) $\quad \dot{\theta}_2 = \frac{\tau}{I_w}t + \dot{\theta}_1$

(5) $\quad t = \frac{I_v}{\tau'}\dot{\alpha}_1$ (6) $\quad t = \frac{I_v\dot{\alpha}_1}{\tau'} \pm \sqrt{\left(\frac{I_v\dot{\alpha}_1}{\tau'}\right)^2 + \frac{2I_v}{\tau'}(\alpha_2-\alpha_1)}$

The equations (1) through (4) are the ordinary equations for uniformly accelerated motion. (5) is a solution of (2) if $\dot{\alpha}_z = 0$, and (6) is a solution of (1).

The solution procedure is as follows. It is assumed that $\dot{\alpha}_z = 0$ and t is computed from (5). Then this value of t is used in (1) to find α_z, while (3) is used to find θ_z. This point is plotted on the α vs θ curve (see Fig. 5, R_1) and from its position with reference to the curve α vs θ we can decide whether or not a collision actually took place <u>after</u> the verge came to rest. If it did not then the procedure for determining the point of collision is exactly as used in the previous work (See R_1 - appendix A) where equation (6) must be used. If, on the other hand, the collision does take place after the verge has come to rest than we know at once that the collision angle θ can be taken from the curve and the known value of α. Next we use (3) to solve for the time t at which the collision occurs and from this get $\dot{\theta}_z$. Similar methods are used in the case of a leading collision.

There is a maximum frictional torque which can be applied without causing the mechanism to jam. At the beginning of the mechanism motion, we assume that the verge is at rest in the equilibrium position. Using the first (leading) equation of page 8, with $\dot{\theta}_o = 0$, we can see that if $\tau(\theta - \theta_a) = \tau(\alpha - \alpha_o)$ there will be no motion. Substitution of these values shows that when $\tau' = 1.1 \tau$ the apparatus will not run.

We calculated the value of $\bar{\dot{\theta}}$ for a number of values of τ' extending from 0 to 2.8 x 10^6 dyne cm, with $\tau = 2.71$ x 10^6 dyne cm. These results are plotted in a graph, Fig. 3. The velocity decreases as τ' increases up to a value $\tau' = 2.4$ x 10^6 after which it begins to increase. At about 5 x 10^6 the velocity would drop rapidly to zero. It can be seen from the graph that the total change in velocity is about 17%.

It is easy to determine also the angular amplitude of the verge, for each friction torque. This was done and the values are given on Fig. 4.
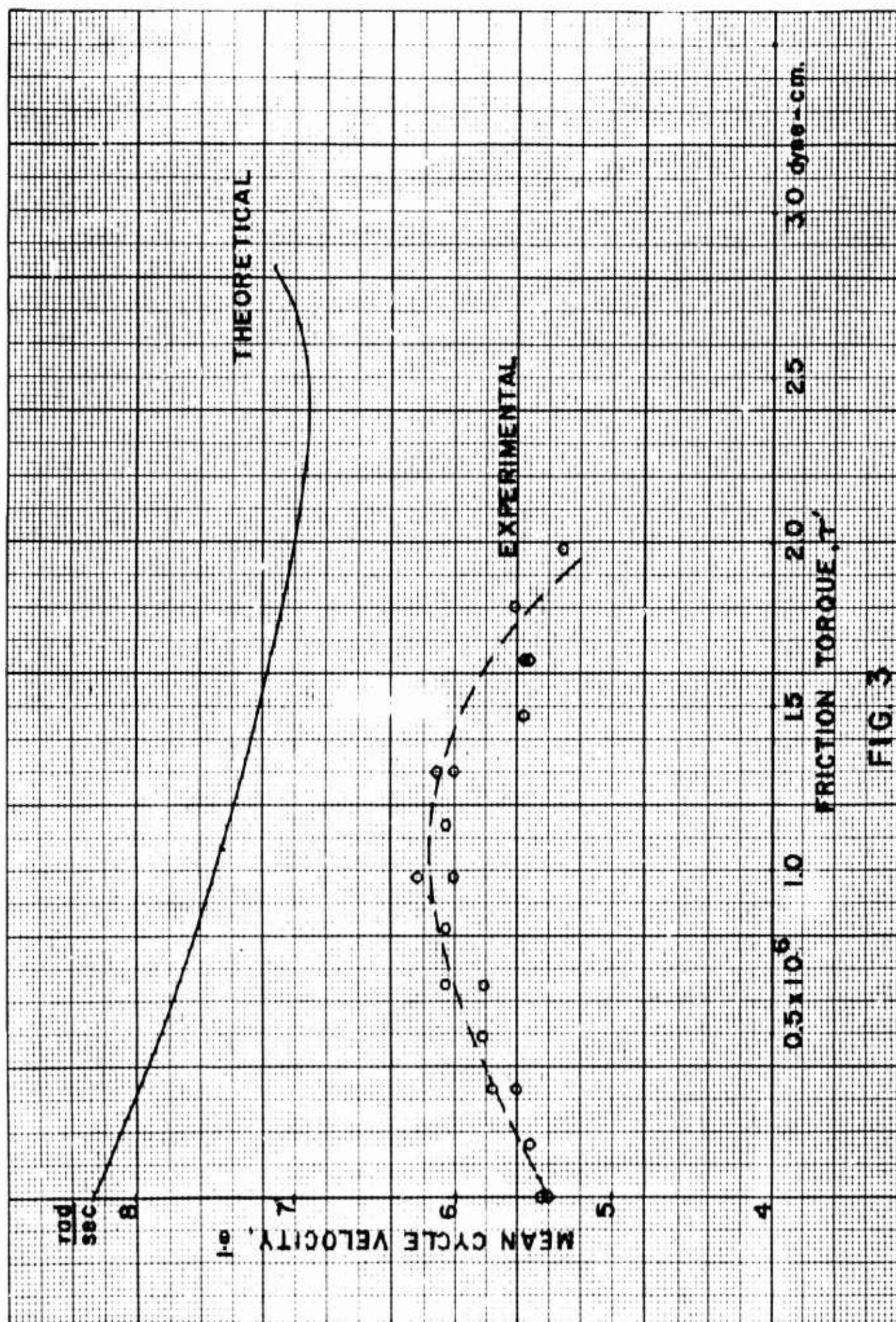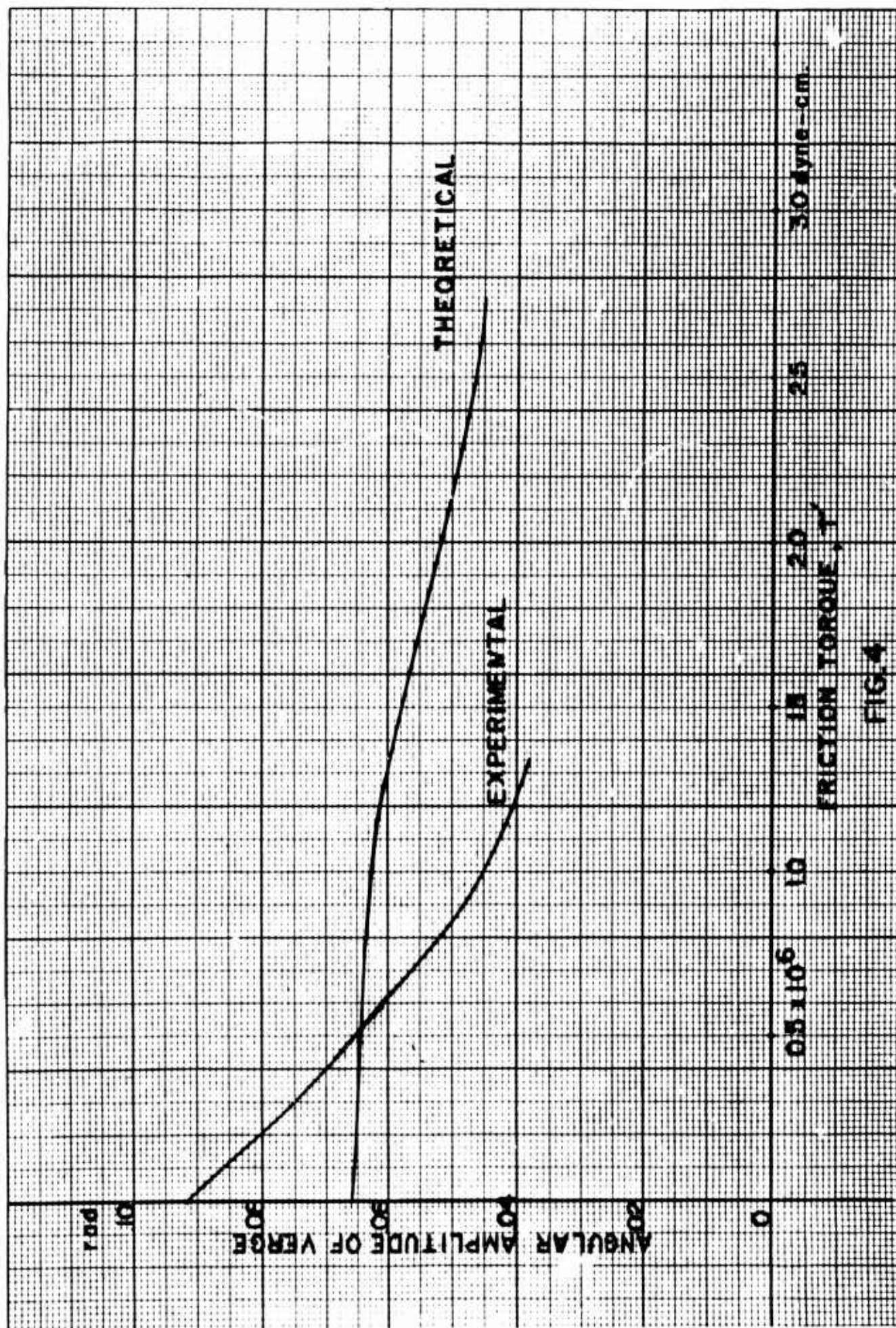
FIG. 3

FIG. 4

VI. COMPARISON OF FRICTIONAL EFFECT - THEORETICAL vs EXPERIMENTAL

When one comes to the point of determining the experimental values of velocity at various values of μ', the results are most erratic. To produce a frictional torque the apparatus described in R_1 - VIII(a) is modified as follows. The horizontal shaft of the verge is extended and there is fastened to it a smooth brass disc. This disc is located between two brass plates one fixed and one movable. These two plates can be made to squeeze the brass disc tightly by hanging weights on a string. The accompanying Figure 5 shows this setup more
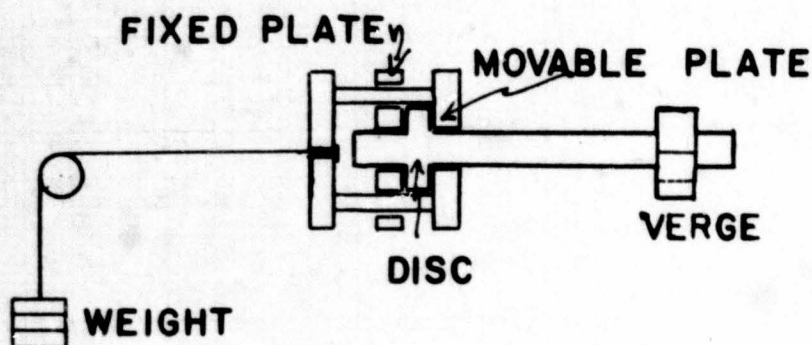

FIG. 5

clearly. Measurement showed that the friction torque on the disc was proportional to the mass of the weights. Also for a frictional torque of about 2×10^6 dyne cm the mechanism jams. This can be compared with the theoretical value of 3×10^6 mentioned above.

Several series of readings were made, but the results were very erratic, and could not be repeated. The general behaviour was as follows. The velocity of the mechanism first increased and then decreased as the load increased. In all cases the maximum change in velocity observed before jamming was about 10%. However, numerical values could not be repeated from one run to the next. The results of two fairly consistent series of readings are plotted on the graph of Figure 3.

One other set of data was obtained experimentally, the angular amplitude of the verge. This value showed a steady decrease as the driving torque was increased. These data are plotted on Figure 4, as are the theoretical values.

One notices at once that the experimental and theoretical results contradict each other for the first instance in this whole study. It would seem important to resolve this contradiction, if possible. To do this first requires a closer analysis of the behaviour of the motion.

During each free period, it is clear that the verge will move more and more slowly as the friction torque increases. Hence the wheel must turn

through a greater and greater angle before collision occurs and so the free period increases while the contact period decreases. This of course is one way of saying that the angular amplitude of the verge is decreasing as friction torque increases. In the limit the value of this angular amplitude would become .0578 radians, the angular distance between the two positions of last contact. During the contact periods one will find that the velocities are decreased because of the opposition of the friction torques. Hence there are two opposing tendencies in these friction torque cycles, (1) a lengthening of the free period which tends to speed up the motion, (2) a slowing down during the contact motion due to friction torque. It is not possible to predict which is the overriding tendency in a qualitative way. On Figure 6 are plotted five curves which afford some quantitative information. Curves 1 - 4 are, as labeled, the times for each of the four portions of the cycle, i.e., leading and trailing free motion, leading and trailing contact motion. Curve 5 is a plot of the total time for the cycle. One immediately observes the striking resemblance in shape between Curves 4 and 5. This indicates most emphatically that by far the most sensitive part of any cycle is leading contact. Below the value $\tau' = 2.4 \times 10^6$ dyne cm, the effect of the decreased velocity of the wheel on contact is the greater. As τ' increases above this value the contact period becomes so small and the free period so large that the mechanism is scarcely slowed down during the contact period. Now we find the velocity increasing again, or the time decreasing.

It should be clear that any factor that would increase the duration of the free period at a given friction torque would result in an increased velocity or decreased time. Now either because the tolerances of the model are great, or, more likely, because teeth and pallet faces are worn, the angular amplitude does decrease much more rapidly in the experimental than the theoretical case. Hence it seems reasonable that this rapid decrease in angular amplitude may overbalance the slowing down effect of the contact motion, so that the net velocity increases up to the jamming point. Within the rather wide limits of uncertainity we found experimentally, this appears to be what happens.
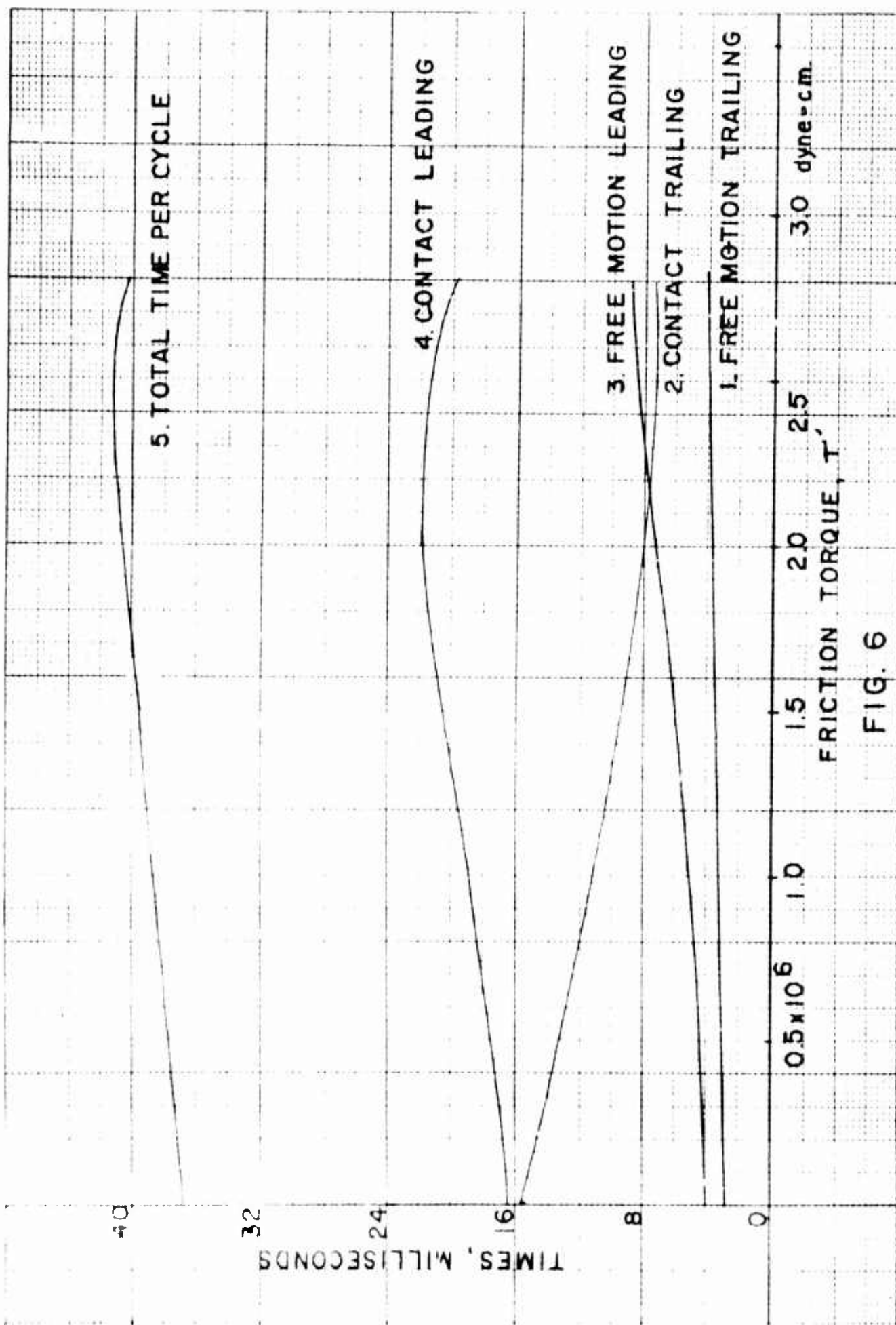
FIG. 6

5. TOTAL TIME PER CYCLE

4. CONTACT LEADING

3. FREE MOTION LEADING

2. CONTACT TRAILING

1. FREE MOTION TRAILING

FRICTION TORQUE, T'

0.5×10⁶   1.0   1.5   2.0   2.5   3.0 dyne-cm.

TIMES, MILLISECONDS

0   8   16   24   32   40

VII. THE EFFECT OF FRICTION TORQUE ON THE ACTUAL MECHANISM.

If a mechanism is mounted in a low g centrifuge and rotated, friction torques will be applied to the mechanism not only at the verge shaft but at all other shafts as well. By changing the speed of the centrifuge the friction torque can be varied. The mechanism was mounted on the centrifuge disc in two ways (1) verge shaft horizontal (2) verge shaft vertical. In the first case, the centripetal force will press the shoulder of the shaft up against the casing, and in the second case it is the cylindrical surface of the shaft which rubs on the bearing hole in the case.
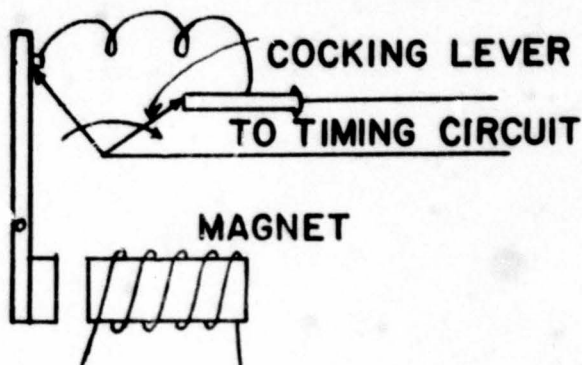
FIG. 7

The mechanism was timed while under the influence of the centrifugal field, using the apparatus shown in Fig. 2, R_3 - II. The mechanism is placed on a stand, which is fastened to the centrifuge disc. The mechanism is cocked, and the centrifuge put into rotation. The turntable is provided with six terminals which are brought out thru brushes to external terminals. Two of these are utilized to carry a current which actuates an electromagnet. This electromagnet serves to release the mechanism from its cocked position.

Two other terminals provide the timing circuit. By these means the mechanism can be made to run while under the influence of the centrifugal field. Details of the electromagnetic release are shown in Figure 7.

The presence of the release lever puts an upper limit on the rotational speed at which the centrifuge can be run. This was such as to make the centrifugal acceleration about 60 g. Above this speed, a component of centripetal force was so directed as to release the mechanism and allow it to run.

A selection of 15 mechanisms was chosen for test and each was timed at values of acceleration extending from accelerations of zero to those of about 60 g. Each mechanism was run in both positions (1) and (2). Now in position (1), the verge shaft horizontal, the axis of the driving spring is also horizontal. When the mechanism is mounted on the disc of the centrifuge, the centrifugal force is so directed as to push the turns of the coil tightly against each other. This of course increases the friction between turns and apparently decreases the effective driving torque since, as the centrifugal force increases, all but one of the mechanisms show an increased period, or decreased velocity.

The same 15 mechanisms were then run in position (2). In this position the axis of the driving spring is vertical, and the centrifugal field will not in this case increase the friction between turns of the spring. In contrast to the results of the first set of measurements, we find that for verge shafts vertical the running time decreases as the centrifugal acceleration increases, or the

velocity of the mechanism increases. There are no exceptions to this be-
haviour. This is of course the behaviour observed experimentally in the
case of the scaled up model, although our apparatus did not allow us to
run the centrifuge at speeds matching the higher friction torques obtained
for the model. On Figure 8 we have plotted mechanism period vs acceleration
(friction torque) for three particular mechanisms of normal behaviour.

The largest variation in running time obtained in case (1) above
was 9% from rest position to a field of 60 g. The mean value of the ob-
served change is about 4% for the sample of 15 mechanisms used. For case
(2) the maximum change is 7% and the mean is 5%. We cannot say what the
total change might be before jamming of the apparatus due to large field
because of the difficulties with the release mechanism already described.
It would seem reasonable to expect a maximum change of no more than 10%.
In closing this discussion we should point out that the friction effect
between turns of the spring is perhaps not the only reason for the dif-
ference in behaviour of cases (1) and (2). It does seem likely, however,
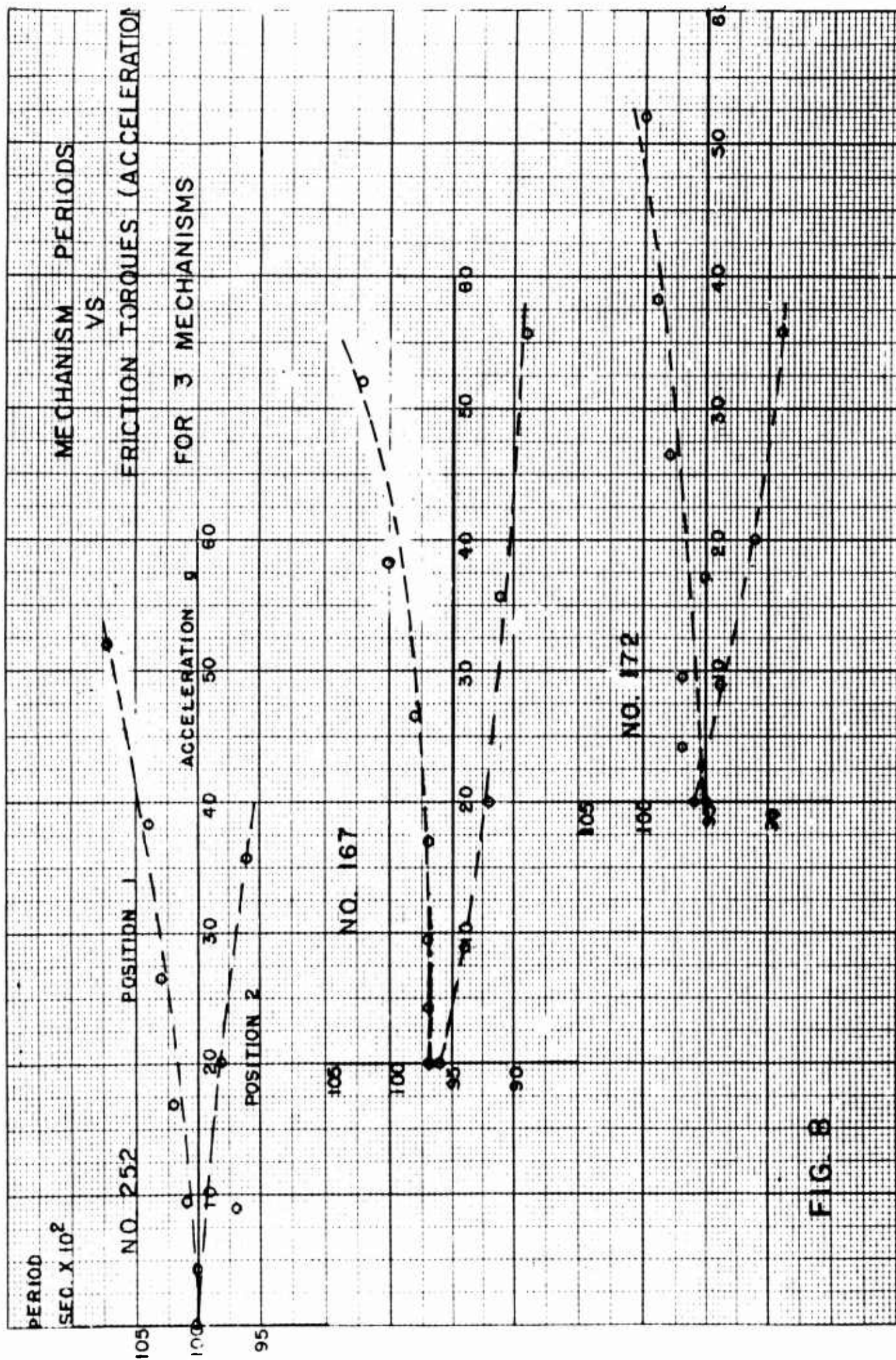to be the most important.

MECHANISM PERIODS VS FRICTION TORQUES (ACCELERATION) FOR 3 MECHANISMS

FIG. 8

## VIII THE EFFECT OF VERGE-WHEEL MATERIAL ON RUNNING TIMES

In all experimental work of the first three reports the model verge and wheel were made of brass, the material of which the actual mechanism is constructed. Our theoretical discussion is incapable of distinguishing between materials. Hence only by experimental measurements can we investigate any effect produced by changing the material of which the verge and wheel are made.

We obtained from the shops of Diamond Ordnance Fuze Laboratories, 10 verge-wheel combinations scaled up 10 times over the actual mechanism. There were two wheel-verge combinations of each of the following materials: brass, aluminum, magnesium, cold rolled steel, and steel (Simmons gr. stock.). A series of trials were made to determine the mean velocity for each of four different verge moments of inertia, and this was done for each of the five kinds of materials. The results are given in graphical form on Fig. 9. Here are plotted $\log \bar{\sigma}$ vs. $\log I_V$. for each of the five cases. These curves do show a difference in running rates among the various materials. Specific conclusions are difficult to draw but it is seen that in general the magnesium mechanism runs most slowly, the steel most rapidly. The cold rolled steel is very little different, on the other hand, than magnesium. In each case except for cold rolled steel, the four points define with fair accuracy a straight line. The experimental measurements were repeated several times but no essential change in results was noted. There is roughly a 10% overall variation among velocities at any one verge moment. Putting it another way the velocities of steel are about 5% greater than those for brass, and the velocities of magnesium about 5% below.

The observed differences in running rates are due partly to slight differences in geometry similiar to those discussed in R_z. Probably the major factors accounting for this difference were the sliding friction between verge face and wheel tooth, and behaviour at collision. The first factor is so variable from one trial to another and one model to another as to defy analysis. In an attempt to learn if there is a difference in the kind of collision which occurs for different materials the Diamond Ordnance Fuze Laboratories made high speed motion pictures of the various verge-wheel combinations in motion for any one material should be identical, but practically the pictures show that this is not true. There is a greater difference between two consecutive cycles of a given verge wheel combination than one can observe between cycles for combinations of different materials.

The same sort of observations were next made on verge wheel combinations of the same five materials as above when verge and wheel are actual size. Measurements of mechanism period were obtained (for each kind of material) for three different verge moments of inertia. The values were obtained by means of the apparatus described in R_3-IV. It is to be remembered that this is merely a mechanism in which are substituted the verge-wheel components under test, and that torque delivered to the star-wheel is the same in all cases. The results are plotted in Figure 10.

LOG $\bar{\theta}$ VS LOG I_v FOR VARIOUS MATERIALS

FIG. 9

LOG X vs LOG I_v FOR VARIOUS MATERIALS

FIG. 10
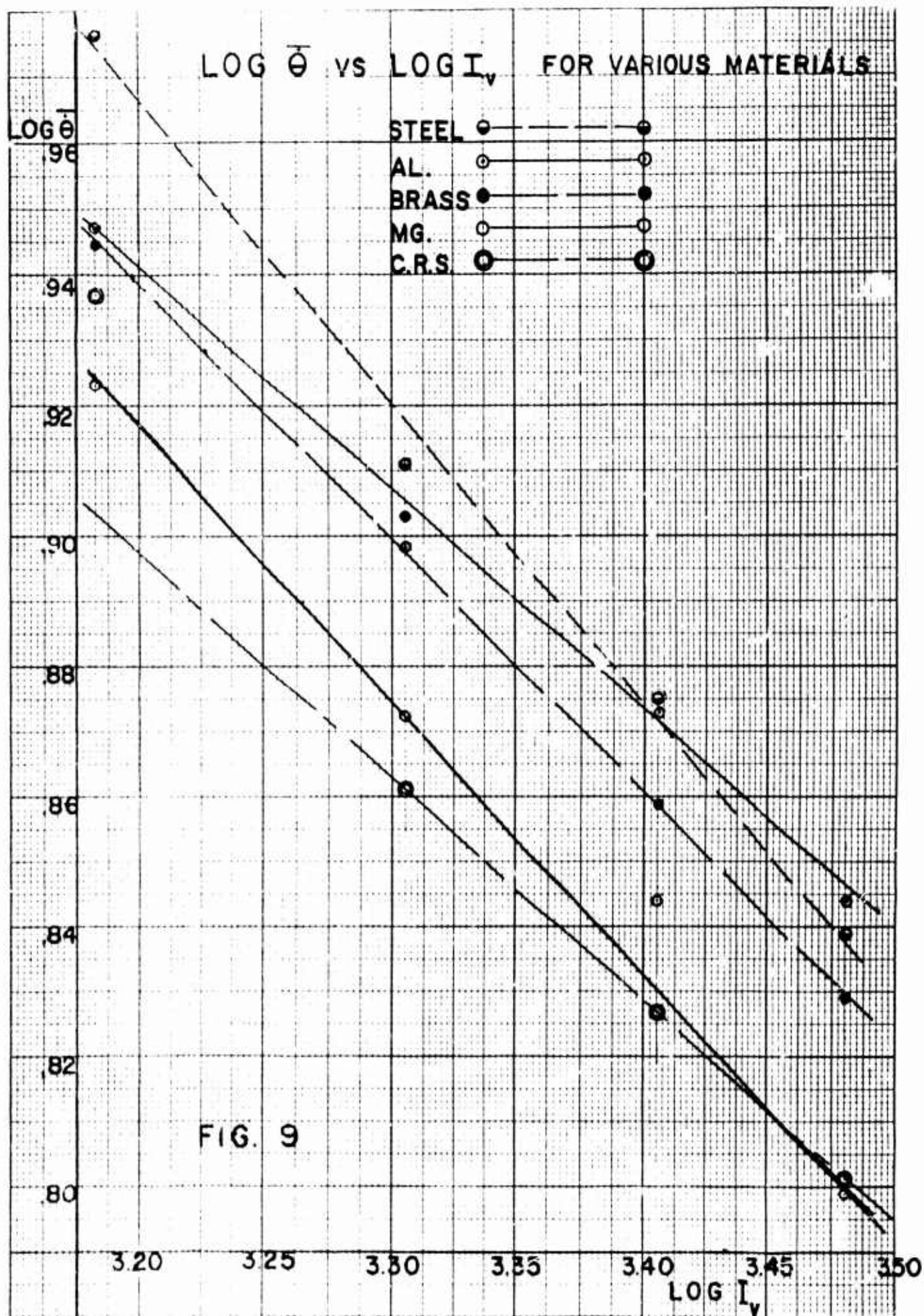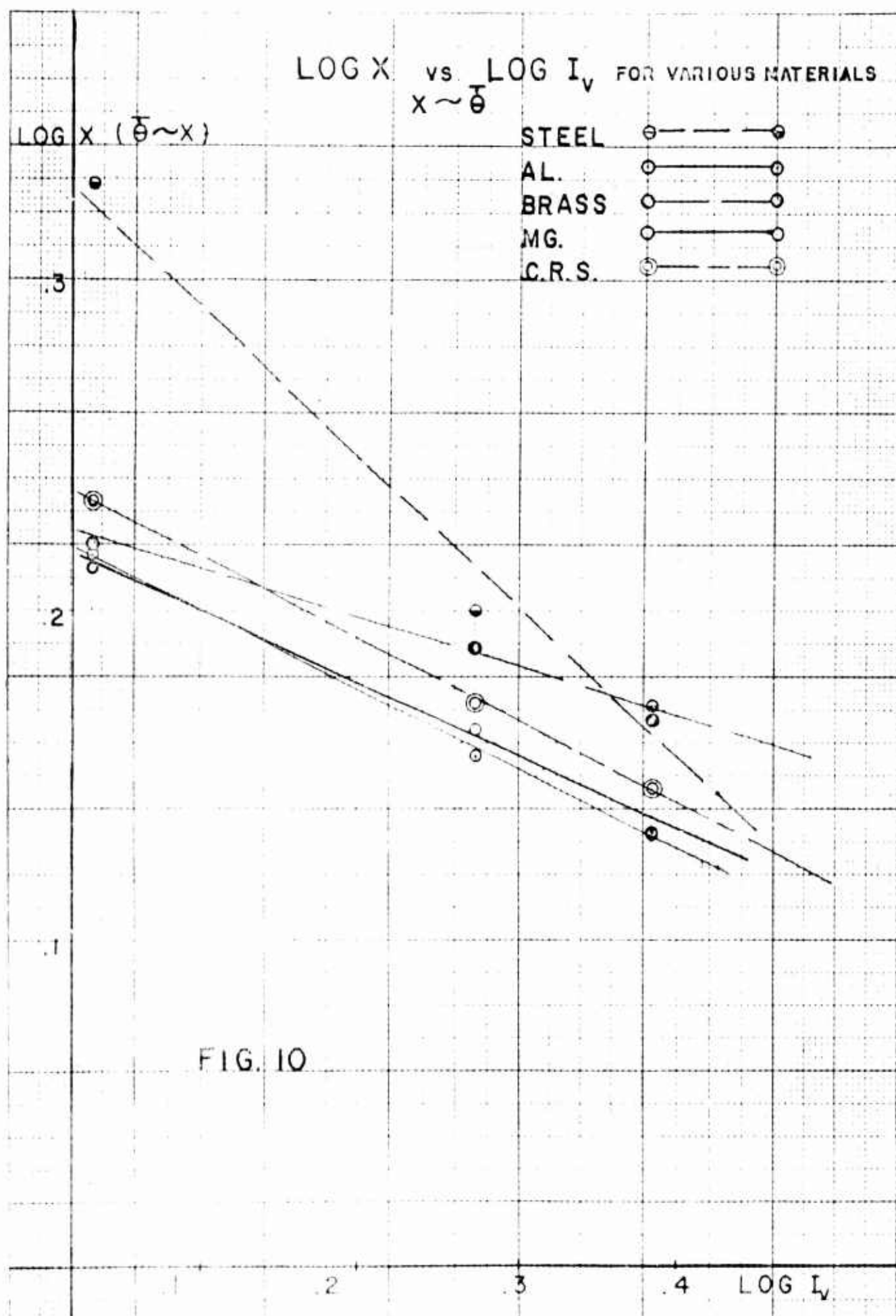
It is not easy to understand the significance of these results which are quite different from those found in the case of the model. There is again a difference in the velocities for the various materials. One cannot correlate the results for mechanisms and models, however, as we see at once by comparing the curves of Figures 9 and 10. In fact, the mechanism results indicate less spread in velocities than we found for the model. (Note that in Figure 10 the plotted ordinates are only _proportional_ to the velocities.) Yet there _is_ a real difference in velocities among verges of a given moment of inertia but different materials.

From R_3 we know how sensitive to changes in geometry these velocities are, and it seems reasonable to expect such an effect here. Close examination of the several verges did not reveal any great departures from standard dimensions, any more, at least, than we found in any sampling of a number of verges. These verges had been filed before they were received but the filing did not change the critical dimensions to any great extent.

On the basis of our results one can conclude only that the material of construction will influence the running rate of a mechanism. How much, or in what direction, our scanty information does not allow us to predict.